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ABSTRACT

The evolution of forces acting on horizontal cylinders subjected to impact by a sinusoidally oscillating free surface was investigated both theoretically and experimentally. The experiments were conducted in a large U-shaped tunnel, with cylinders 3 to 8 in. (76 to 203 mm) in diameter. The results are expressed in terms of three force coefficients: (1) the general slamming coefficient that expresses the normalized force acting on the cylinder at any time after the impact, (2) the normalized impact force at the initial instants of slamming, and (3) the maximum drag coefficient that occurs when the cylinder is immersed in water about 1.8 diameters. The slamming-force coefficient was found to equal 3.2. Also, the force experienced by the cylinder cannot be considered independently of the dynamic response of that cylinder. In fact, the slamming-force coefficient may be amplified to a value as high as 6.3 through the dynamic response of the cylinder and its supports.

INTRODUCTION

Information about the forces acting on bluff bodies subjected to wave slamming is of significant importance in ocean engineering and naval architecture. The design of structures that must survive in a wave environment depends on a knowledge of the forces that occur at impact, as well as on the dynamic response of the system. Two typical examples include the structural members of offshore drilling platforms at the splash zone and the often encountered slamming of ships.

The general problem of hydrodynamic impact has been studied extensively, motivated in part by its importance in ordnance and missile technology.[1] Extensive mathematical models have been developed for cases of simple geometry, such as spheres and wedges. These models have been well supported by experiment. Unfortunately, the special case of wave impact has not been studied extensively. Kaplan and Silbert[2] developed a solution for the forces acting on a cylinder from the instant of impact to full immersion. Dalton and Nash[3] conducted slamming experiments with a 0.5-in. (12.7-mm) diameter cylinder and small amplitude waves created in a laboratory tank. Their data exhibited large scatter and showed no particular correlation with either the predictions of the hydrodynamic theory or identifiable wave parameters. Miller[4] presented the results of a series of wave-tank experiments to establish the magnitude of the wave-force slamming coefficient for a horizontal circular cylinder. He found an average slamming coefficient of 3.6 for those trials in which slamming was dominant.

Evaluating slamming effects with wavy flows is extremely difficult partly because of the limited range of wave amplitudes that can be achieved and partly because of the difficulty of measuring the fluid velocities at the instant of impact.

Faltinsen *et al.*[5] investigated the load acting on rigid horizontal circular cylinders (with end plates and length-to-diameter ratios of about 1) that were forced with constant velocity through an initially calm free surface. They found that the slamming coefficient ranged from 4.1 to 6.4. They also conducted experiments with flexible horizontal cylinders and found that the analytically predicted values were always lower (50 to 90%) than those found experimentally.

This investigation was undertaken (1) to examine the existing theoretical models for determining wave slam forces on circular cylinders; (2) to furnish data, obtained under controlled laboratory conditions, about forces acting on circular cylinders subjected to impact with a sinusoidally oscillating water surface; (3) to determine the relative importance of the inertia- and drag-dominated forces during fluid impact; and (4) to correlate these data for identifiable wave parameters such as the Froude number (N_{Fr}); the Keulegan-Carpenter number (N_K); and the Reynolds number (N_{Re}).

This investigation does not deal with the relatively more complex impact situations arising from the slamming of random ocean waves on the members of offshore structures. Such practical problems require not only a clear understanding of the origins and magnitude of the impact forces through idealized experiments, but also the use of statistical design methods and some knowledge of

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the probabilities associated with various force and response coefficients.

THEORETICAL ANALYSIS

The traditional approach used to design offshore structures involves the classical Morison equation for determining the forces caused by wave motion. Theoretical and practical aspects of the determination of wave forces on offshore structures were described in detail by Sarpkaya and Isaacson recently.[6]

Wave impact generally is defined as the early stages of the penetration of a solid body into a wave surface. At the instant of contact, the fluid near the body undergoes large accelerations that create large forces. As the body becomes more fully immersed, forces resulting from viscous drag and separation effects dominate. Thus, the inertia and drag coefficients used in Morison's equation are not constant, and the problem becomes very difficult, even for simple geometries.

The general case of hydrodynamic impact usually is described by using incompressible potential flow theory. For a moving body with mass, M, and velocity, v_o, impacting a quiescent surface, the system momentum is Mv_o. Neglecting nonconservative forces, the momentum of the system is unchanged during penetration. However, the mass of the system increases because of the fluid set in motion near the body. Also known as added mass, m results in reducing the velocity. Thus, the system momentum after penetration is $(M + m)v = Mv_o$. The impact force at any instant is a function of m and $\partial m/\partial t$. Therefore, the solution requires knowledge of the added mass and its time derivative.

As shown by Kaplan and Silbert,[2] the force acting on a horizontal circular cylinder by a wave system that propagates normal to it is equal to the sum of the buoyant force and the time-rate of change of momentum. Thus, one has

$$\frac{F}{L} = \rho g A_i + (m + A_i)\,\ddot{\eta} + \frac{\partial m}{\partial z}\,\dot{\eta}^2 . \qquad (1)$$

in which F represents the force acting on the cylinder; L, the length of the cylinder; ρ, the fluid density; g, the gravitational acceleration; A_i, the immersed area; m, the added mass per unit length; η, the instantaneous height of the wave surface above the mean water level; and z, the instantaneous depth of cylinder immersion. The first and second derivatives of η for time are denoted by $\dot{\eta}$ and $\ddot{\eta}$. The added mass is given by Taylor[7] as

$$m = \frac{1}{2}\rho r^2 \left[\frac{2\pi^3}{3}\,\frac{(1-\cos\theta)}{(2\pi-\theta)^2} + \frac{\pi}{3}(1-\cos\theta) + \sin\theta - \theta\right] \qquad (2)$$

in which r represents the radius of the cylinder, and θ is defined as shown in Fig. 1.

The motion of the free surface, η, is related to the maximum amplitude by

$$\eta = A \sin 2\pi t/T , \qquad (3)$$

where A and T represent the amplitude and period of the free-surface oscillations. Eq. 1 also can be written in the form of a slamming coefficient, $C_s = 2F/\rho DLU_m^2$, as

$$C_s = \bar{A}_i \frac{gr}{U_m^2} - (\bar{m} + \bar{A}_i)\frac{r}{A}\sin\frac{2\pi}{T}t + \frac{\partial \bar{m}}{\partial \bar{z}}\cos^2\frac{2\pi}{T}t , \qquad (4)$$

where

$$\bar{A}_i = A_i/r^2 , \quad \bar{m} = m/\pi r^2 ,$$

$$\bar{z} = z/r , \quad U_m = 2\pi A/T$$

With $\bar{z} = r[1 - \cos(\theta/2)]$, Eq. 2 can relate the added mass to the depth of immersion. Clearly, $\partial\bar{m}/\partial\bar{z}$ begins with an initial value of π and drops rapidly. The quantity gr/U_m^2 is related to the Froude number by $N_{Fr} = U_m^2/2gr$. Thus, $C_s = f(N_{Fr}, r/A, \partial\bar{m}/\partial\bar{z})$.

The rate of change of the normalized added mass with $\bar{z}$ depends on θ and, hence, on the time measured from the instant of impact. For example, for the case where $\eta_o = 0$ and $t = 0$, $\partial\bar{m}/\partial\bar{z} = \pi$, and $C_s^o = \pi$. Consequently, for the particular case under consideration, C_s^o at the instant of impact does not depend on either the size of the cylinder or the flow parameters.

Because the cylindrical members of a structure in the splash zone are not necessarily located at the mean water level requires the determination of the particular value of η_o for which there is maximum slamming force at the time of impact. Using Eq. 4, it can be demonstrated that the maximum impact force occurs for the case of $\eta_o = 0$. For this purpose, Eq. 4 was evaluated with the aid

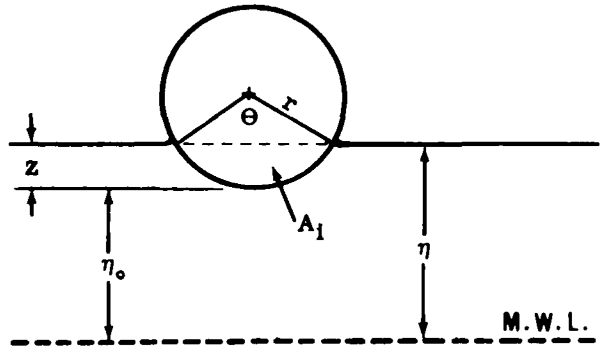

FIG. 1 — DEFINITION SKETCH FOR THE THEORETICAL ANALYSIS.

of a computer that allowed variations in r/A and η_o/A. C_s then was plotted as a function of z/D from zero to unity. Fig. 2 is a typical example of one of these plots. The particular values used were $r/A = 0.125$, and $\eta_o/A = 0.0$, 0.40, and 0.80. As can be seen readily from Fig. 2, C_s is largest at $z/D = 0$ for $\eta_o/A = 0$, starts at a value of π, and drops to a minimum at z/D of about 0.5. C_s rises again as z/D approaches unity. Thus, Eq. 4 indicates that C_s, and consequently the impact force, is of an implusive nature beginning with a finite value at the instant of impact. Since viscous forces are neglected, one would expect the solution to deviate from the actual situation as the cylinder becomes more fully immersed. Where this becomes the case can be determined only experimentally.

However, one can assume that the solution would be valid for small depths of immersion, which is the case for the instant of impact up to z/D of 0.02 or more. With this restriction, $\bar{A}_i$ and $\bar{m}$ are small, as is $\sin 2\pi t/T$. Also, $\cos^2 2\pi t/T$ is almost equal to 1.0. Therefore, C_s reduces to

$$C_s \simeq C_s^o = (\partial \bar{m}/\partial \bar{z})_{t=0} = \pi \cdot \quad \ldots (5)$$

It is unrealistic to assume that the impact force rises from zero to π instantaneously. Several factors, specifically the compressibility of the air between the cylinder and water surface, entrapped gases in the water, and surface irregularities, would account for some finite rise time. Nonetheless, the rise time can be expected to be short — i.e., in the order of milliseconds. The exact nature of the rise is an interesting question for further study. However, here C_s is assumed to vary linearly during a rise period, t_r. For $t > t_r$, C_s drops from π as $\partial\bar{m}/\partial\bar{z}$. Fig. 3 is a representation of this assumption. Exactly how long a time interval t_r is will be discussed later.

The realization that the impact force has an implusive nature requires consideration that this force does not act on a perfectly rigid body, but rather on a cylinder that is supported elastically. The response of such a system approaches that of a rigid body only if its natural frequency approaches infinity. Additionally, the response of the system to an impulsive force depends heavily on the exact nature of the force itself as well as on the system natural frequency.

In general, the instantaneous displacement, $x(t)$, of a system of mass M with a spring constant k is given by (Ref. 8)

$$x(t) = \int_0^t F(\xi)\, g(t-\xi)\, d\xi \;, \quad \ldots (6)$$

in which ξ represents a dummy variable; $F(t)$, the driving force; and $g(t)$, the response to a unit step excitation. Eq. 6 readily is recognized as the Duhamel superposition integral, which can be expanded as follows for $g(t) = (1/M\omega_n) \sin\omega_n t$,

$$x(t) = - \frac{1}{M\omega_n} \cos\omega_n t \int_0^t F(t) \sin {}_n t \, dt + \frac{1}{M\omega_n} \sin\omega_n t \int_0^t F(t) \cos\omega_n t \, dt \quad \ldots (7)$$

in which ω_n is the natural frequency of the cylinder and supports.

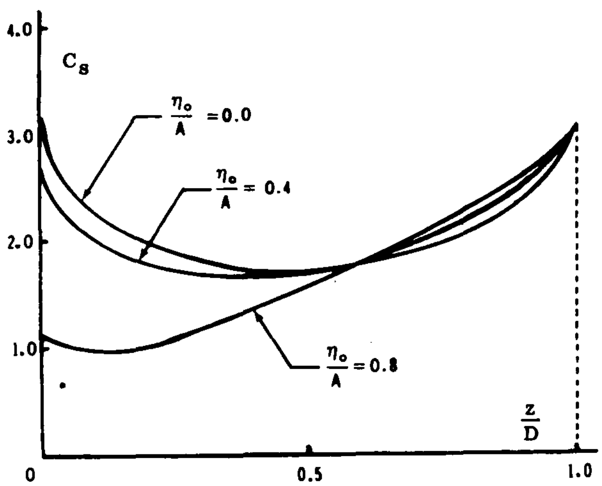

FIG. 2 — THEORETICAL VALUES OF THE SLAMMING COEFFICIENT AS A FUNCTION OF RELATIVE SUBMERGENCE AND POSITION.

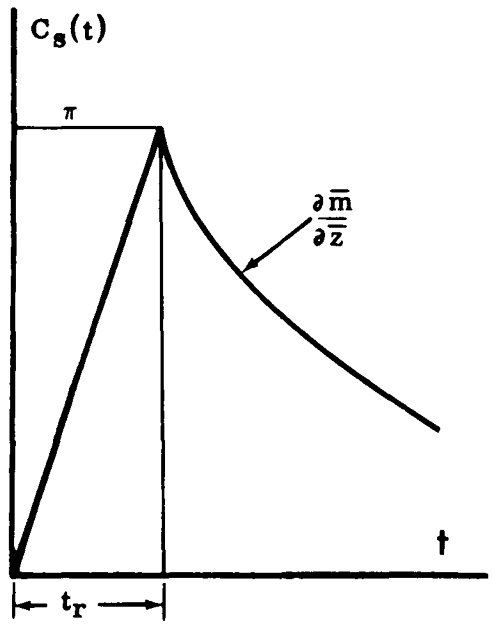

FIG. 3 — RISE TIME AND THE SLAMMING COEFFICIENT AS A FUNCTION OF TIME.

By letting $F(t) = F_o f(t)$ and by changing the variable of integration to $\beta = \omega_n t$, we obtain

$$\frac{k\ x(t)}{F_o} = -\cos\beta \int_0^{\beta/\omega_n} f(t)\sin\beta\, d\beta + \sin\beta \int_0^{\beta/\omega_n} f(t)\cos\beta\, d\beta \quad \ldots\ldots\ldots (8)$$

in which $k = \omega_n^2 M$, the spring constant. The term on the left side can be interpreted as the ratio of the instantaneous force sensed by the supports of the cylinder to the actual force acting on the cylinder. For the simple case in which $F(t)$ is a step function, Eq. 8 reduces to

$$\frac{k\ x(t)}{F_o} = (1 - \cos\omega_n t) \cdot \quad \ldots\ldots (9)$$

It is apparent from Eq. 9 that the instantaneous force sensed by the system can be anywhere from zero to two times the actual mean force.

If one assumes, as before, that the impact force is as shown in Fig. 3, then Eq. 8 must be evaluated using $C_s(t)$ as $F(t)$. Additionally, damping can be accounted for by writing $g(t)$ as

$$g(t) = e^{-\zeta\omega_n t}\sin\omega_n t, \quad \ldots\ldots (10)$$

where ζ is the damping coefficient. Thus, Eq. 8 can be rewritten by replacing the forcing function $F(t)$ by $C_s(\beta)$ as

$$\frac{k\ x(t)}{C_s^o} = -\cos\beta \int_0^{\beta/\omega_n} C_s(\beta)\, e^{-\zeta\beta}\sin\beta\, d\beta + \sin\beta \int_0^{\beta/\omega_n} C_s(\beta)\, e^{-\zeta\beta}\cos\beta\, d\beta \quad \ldots\ldots\ldots (11)$$

Eq. 11 was solved numerically using a trapezoidal integration scheme. Values of ω_n were taken as 358 s^{-1} and 628 s^{-1}, which corresponded to the measured values of ω_n for a 6.0-in. (153-mm) and a 3.0-in. (76-mm) diameter cylinder, respectively. The damping coefficient was 0.014 in both cases, also corresponding to the measured values. The rise time varied from zero to about 0.025 s. Figs. 4 through 7 are representative plots of $kx(t)/C_s^o$ for the 6-in. (153-mm) cylinder for various t_r.

Fig. 4 represents Eq. 11 plotted for a rise time of 0.0001 s; Fig. 5 represents a rise time of 0.0100 s; Fig. 6 is a rise time of 0.0195 s; and Fig. 7 represents a rise time of 0.0230 s. Note that for a very short rise time, the value of $kx(t)/C_s^o$ reaches a value of about 1.7 at $t = 0.001$ s and then drops off, with a "double peak" appearing at a rise time of 0.0195 s.

The interpretation of these results indicates that, depending on the rise time, values of the apparent slamming coefficient, C_s^o, may lie between roughly 0.5 and 1.7 of the theoretical value of π. Again, this applies only to a small depth of immersion corresponding to the initial moments of the impact. The significance of these facts is that values of C_s determined experimentally from the measured reaction forces at the supports of the cylinder may show wide scatter, depending on the dynamic response of the cylinder and the rise time. Also, if the surface is not perfectly plane, rise times may vary from experiment to experiment, resulting in an apparent nonrepeatability. This is particularly true for conditions in the ocean environment. Evidently, controlled laboratory experiments help to establish the ideal value of the

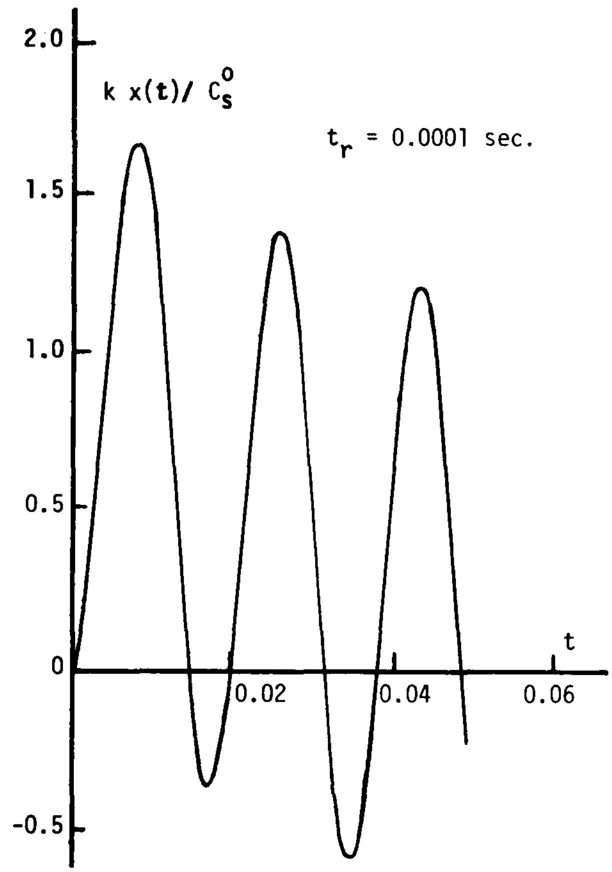

FIG. 4 — AMPLIFICATION FACTOR AS A FUNCTION OF TIME FOR A RISE TIME OF 0.0001 s.

impact coefficient and to explain the reasons for the observed scatter in the data. The determination of the impact-force magnification factor for the cylinder in the ocean environment necessarily must consider the random nature of the disturbances at the wave surface, the orientation of the structural member for a given wave, currents, three-dimensional nature of the waves, spray, etc. In short, there are no deterministic means for predicting the reaction forces acting on the supports of a member resulting from wave impact, even when the structural characteristics of the member (damping, natural frequency, etc.) and the ideal value of the impact force are known. This is true because the rise time depends on all these nondeterministic conditions.

EXPERIMENTAL EQUIPMENT AND PROCEDURES

The equipment used to create the harmonically oscillating flow has been used extensively at the Naval Postgraduate School for the past 4 years. The apparatus is described in Ref. 9. Only the salient features, most recent modifications, and the adaptation for this study, are described briefly in the next section.

U-SHAPED WATER TUNNEL

The tunnel is 32.8 ft (10 m) long × 20 ft (6 m) high with a 3 × 3 ft (91 × 91 cm) test section. The two vertical legs are 3 × 6 ft (91 × 182 cm) in cross-section. Previously, a butterfly valve arrangement at the top of one leg was used to start the oscillations. Recently, the tunnel was modified so that oscillations could be generated and maintained indefinitely at the desired amplitude. A 2-Hp (1.5-kW) fan was connected to the top of one leg of the tunnel with a 2-ft (60-cm) diameter pipe. A small butterfly valve, placed in a special housing between the top of the tunnel and the 2-ft (60-cm) supply line, oscillated harmonically at a frequency equal to the natural frequency of the oscillations in the tunnel. The oscillation of the valve was synchronized perfectly with that of the flow using a feedback control system. A pressure transducer (sensing the instantaneous acceleration of the flow) was connected to an electronic speed-control unit coupled to a DC motor that oscillated the valve plate. The circuit maintained the period of oscillations of the valve within

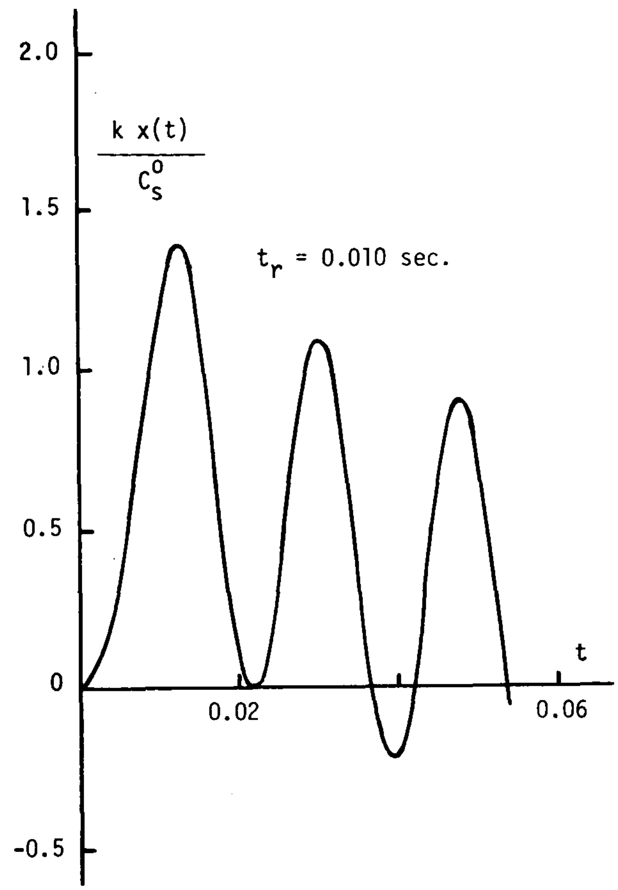

FIG. 5 — AMPLIFICATION FACTOR AS A FUNCTION OF TIME FOR A RISE TIME OF 0.010 s.

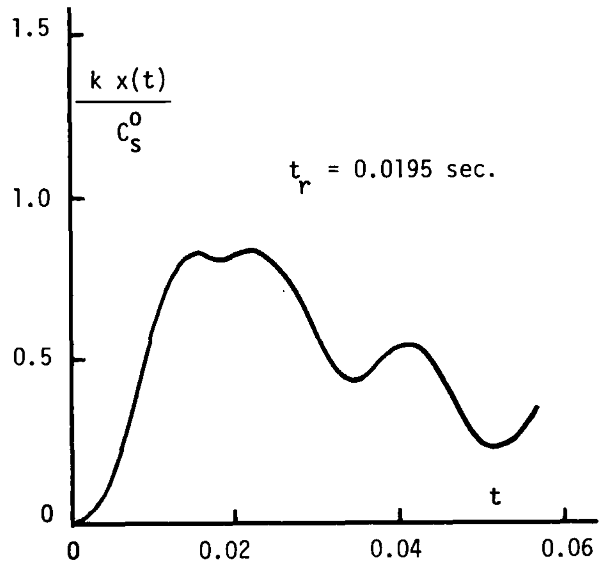

FIG. 6 — AMPLIFICATION FACTOR AS A FUNCTION OF TIME FOR A RISE TIME OF 0.0195 s.

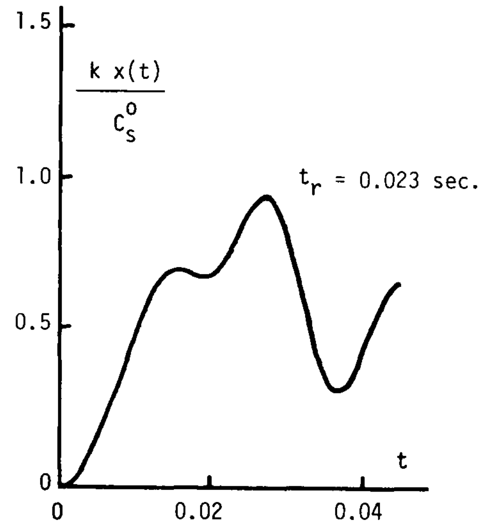

FIG. 7 — AMPLIFICATION FACTOR AS A FUNCTION OF TIME FOR A RISE TIME OF 0.023 s.

±0.001. The fluid oscillated with a period of 6.00 s. The amplitude of the oscillations was varied by constricting or enlarging an orifice at the exit of the fan. The orifice opening was calibrated vs the amplitude of flow oscillation. It was possible to oscillate the flow at the desired amplitude as long as desired.

TEST CYLINDERS

About 3- to 8-in. (76.2- to 203-mm)-diameter smooth and rough aluminum cylinders were used for measuring the impact forces. An accelerometer was placed inside each cylinder to measure the instantaneous vertical acceleration of the cylinder. Caps fitted to each end prevented water leakage into the cylinder. Each cap contained a double-precision ball bearing mounted flush with the face. The force transducers were attached to the cylinder by these bearings.

The cylinders were placed in the 3 × 6 ft (91 × 182 cm) section of one leg of the tunnel. Since the height of the mean water level for the bottom of the cylinder was shown by analysis to be important, the water level in the tunnel was adjusted carefully by slowly filling and emptying the tunnel until a slight ripple occurred at the water surface because of contact with the cylinder. This method assured that the mean water level coincided with the bottom surface of the cylinder. The damping coefficient and the natural frequency of the cylinder were determined in air and water by plucking excitation.

FORCE MEASUREMENTS

Two cantilever-beam force transducers measured the instantaneous in-line (drag) and transverse (lift) forces. The strain gauges attached to the beam emitted electric signals. Calibration of the transducers was accomplished by hanging a load from the center of the cylinder. This not only established the level of the electrical output for a known load, but also reaffirmed that the transducers were positioned correctly.

As with other elements of the experimental apparatus, these transducers had been used here for more than 4 years. The details of their construction may be found in Ref. 9. No changes have been noted in the calibration of these transducers since their installation.

Initial experimental efforts established that the transverse forces were small compared with the in-line forces, and consequently measurement of these forces was discontinued early in the study.

The data were recorded with a high-speed tape and a strip-chart recorder. The strip-chart recorder operated at a speed of 200 mm/s, each division representing 0.005 s. Fig. 8 is a typical example of the data recorded.

REDUCTION OF DATA

Three force coefficients were defined in this investigation. The first two are given by

$$C_s = 2F/(\rho DLU_m^2) \quad \ldots\ldots (12)$$

and

$$C_h = 2F/(\rho DLU_m^2) - gD\pi/(2U_m^2) \quad \ldots\ldots (13)$$

The former refers to the slamming coefficient as defined by Eq. 4 and includes the contribution of the dynamic response of the system. The latter represents the second maximum of the normalized force with the buoyancy subtracted. This maximum occurs after the cylinder is fully immersed. Fig. 9 is a typical plot of the measured value of C_s, the theoretical value of C_s from Eq. 4, and the normalized buoyant force.

The third coefficient is C_s^o, which represents the

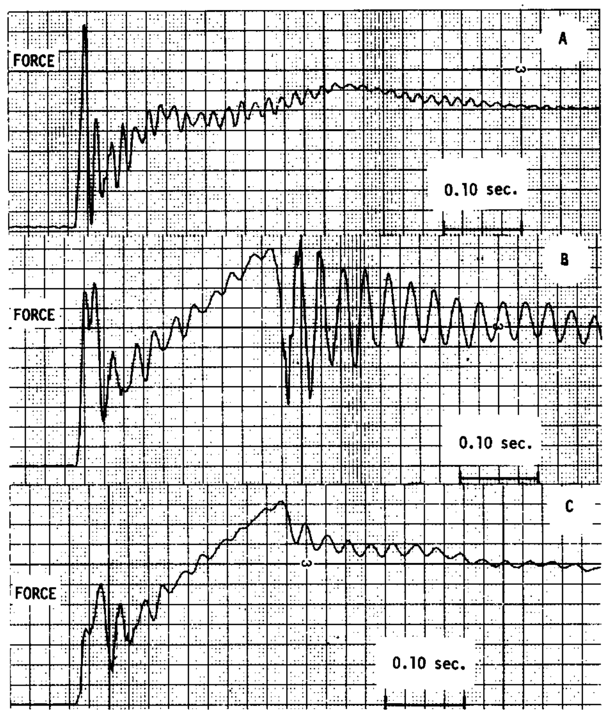

FIG. 8 — SAMPLE DATA TRACES SHOWING THE EVOLUTION OF THE IMPACT FORCE FOR VARIOUS RISE TIMES.

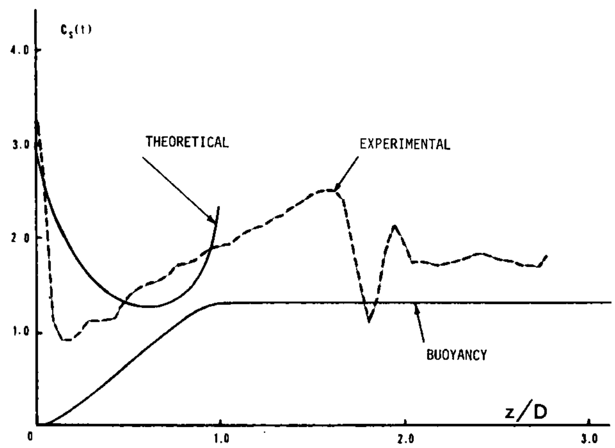

FIG. 9 — EVOLUTION OF THE BUOYANT FORCE AND THE THEORETICAL AND EXPERIMENTAL VALUES OF THE IMPACT FORCE AS A FUNCTION OF RELATIVE SUBMERGENCE.

normalized impact force acting on the cylinder at the initial instants of the impact. This coefficient does not include the contribution of the dynamic response of the cylinder.

The fluid force acting on the cylinder is amplified or attenuated by the dynamic response of the cylinder-transducer assembly. Thus, it was necessary to subtract the inertial force acting on the cylinder assembly from the recorded reaction force. First, the correct relationship between the acceleration of the cylinder and the inertial force acting on the force transducer was established by plucking the cylinder and allowing it to freely vibrate in air. Then, the inertial force corresponding to the instantaneous acceleration was subtracted from the total force sensed by the transducer after the impact of the fluid surface on the cylinder. The net fluid force was normalized as in Eq. 12 to yield the initial value of the slamming coefficient, C_s^o.

DISCUSSION OF THE RESULTS

Experiments with all the cylinders tested for all Reynolds and Froude numbers yielded

$$C_s^o = 3.17 \pm 0.05 \cdot$$

Thus, the force transfer coefficient for slamming at the initial instants of impact is very close to its theoretical value of π. As shown previously, the maximum slamming force may be amplified by a factor of as much as 1.7 by the dynamic response of the structure.

Fig. 10 plots $kx(t)/C_s^o$ as a function of $f_n t_r$ for both the first and the second peak (see Figs. 8b and 8c), using ω_n values for both a 6-in.(153-mm) and a 3-in. (76-mm) cylinder. The drop in the magnification factor for the first peak with increasing rise in time is apparent. Also, the effect of ω_n is confined to a narrow range of $f_n t_r$ values smaller than about 0.5. The force acting on the cylinder is amplified by the dynamic response of the system for a value of $f_n t_r$ smaller than 0.9 and attenuated for $f_n t_r$ values larger than 0.9. The second peak occurs only for $f_n t_r$ values larger than 1.09. The amplification factor for this peak increases from 0.84 to about unity as $f_n t_r$ increases from 1.09 to 1.5.

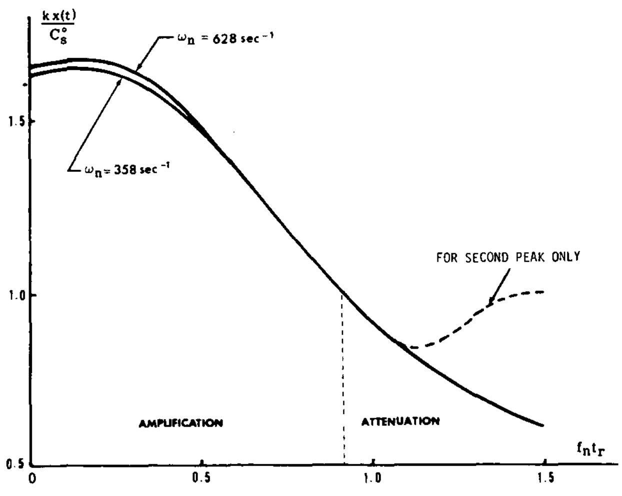

FIG. 10 — AMPLIFICATION FACTOR AS A FUNCTION OF $f_n t_r$ FOR TWO REPRESENTATIVE NATURAL CIRCULAR FREQUENCIES.

A comparison of Figs. 4 and 8a, 6 and 8b, and 7 and 8c shows that the analysis can predict accurately the evolution of the impact force and help explain the differences in the initial value of the impact force, if the rise time or $f_n t_r$ is known.

Following the initial impact, the net force acting on the cylinder begins to decrease, since the contribution of the added mass decreases. During this stage of the flow, the cylinder undergoes damped oscillations at its natural frequency. As the free surface rises further, the buoyant force increases and the separation effects begin to create larger drag forces. The buoyancy-subtracted fluid force reaches its maximum at z/D values from about 1.5 to 2. Even though the present flow situation, in which there is a free surface, cannot be compared directly with an impulsive flow about a circular cylinder, the rise of the drag coefficient to a maximum at $z/D = 1.75$ is much like the rise of the drag coefficient to a maximum at a relative fluid displacement of about 2 in impulsively started flow about a circular cylinder.[10] This rise in the drag coefficient results from the formation of two symmetrical vortexes behind the cylinder. As the motion continues, the vortexes become asymmetrical and shed alternately.

The maximum drag force was corrected for buoyancy, normalized by $0.5\rho DLU_m^2$, and plotted as a function of the Froude number in Fig. 11. Apparently, C_h (see Eq. 13) decreases with increasing Froude number to a value of about unity and remains nearly constant for all Froude numbers from 0.5 to about 2.

Experiments with cylinders having relative roughnesses of 1/100 and 1/50 have shown that the slamming coefficient, C_s^o, remains unchanged. Roughness tended to increase the rise time and decreased the force-amplification factor. Calculations for these coefficients were based on the apparent diameters of the cylinders.

CONCLUSIONS

Theoretical and experimental investigation of

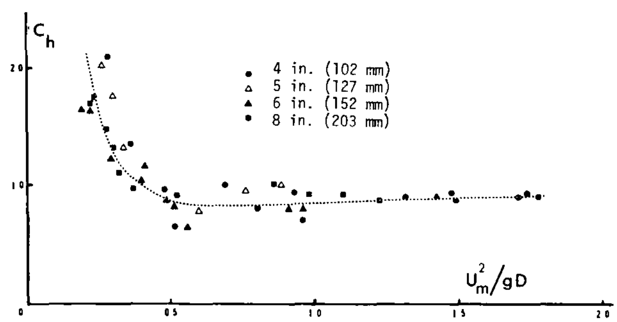

FIG. 11 — DRAG-DOMINATED FORCE COEFFICIENT, C_h, AS A FUNCTION OF THE FROUDE NUMBER.

flow impact on circular cylinders warranted the following conclusions.

1. The dynamic response of the system is as important as the impact force; one cannot be determined without accounting for the other.

2. The initial value of the slamming coefficient is essentially equal to its theoretical value of π. The system response may be amplified or attenuated, depending on its dynamic characteristics.

3. After impact, the cylinder undergoes damped oscillations at its natural frequency.

4. The buoyancy-corrected normalized force in the drag-dominated region reaches a maximum at a relative fluid displacement of about 1.75. Subsequently, the shedding of vortexes and the deceleration of flow reduces the maximum drag coefficient to less than unity.

5. Since the rise time is not deterministic, the slamming-force amplification or the dynamic response of the system should be analyzed using the theoretical value of the slamming coefficient and a rise-time distribution function.

6. Roughness increases the rise time and tends to decrease the amplification factor.

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